HEAT TRANSFER COEFFICIENTS OF STEAM WATER MIXTURES

LIONELL H. NELSON

HEAT TRANSFER COEFFICIENTS

OF

STEAM WATER MIXTURES

* * * *

L. H. Nelson

HEAT TRANSFER COEFFICIENTS

OF

STEAM WATER MIXTURES

by

Lionell Herbert Nelson

Lieutenant, United States Navy

Submitted in partial fulfillment
of the requirements
for the degree of
MASTER OF SCIENCE
IN
MECHANICAL ENGINEERING

United States Naval Postgraduate School
Monterey, California

1 9 5 5

This work is accepted as fulfilling

the thesis requirements for the degree of

MASTER OF SCIENCE

IN

MECHANICAL ENGINEERING

from the

United States Naval Postgraduate School

PREFACE

With power demands increasing and natural power sources decreasing it becomes more and more essential to attain maximum efficiency in the conversion of fuel into useable power. The advent of controlled nuclear power, with its present and future applications to industrial use and as a source of propulsion power, has again stressed the fact that steam will, in all probability, remain the primary means of transmitting heat energy into useful work.

In view of the foregoing, the attainment of higher thermal efficiencies in the use of steam and water as transfer media becomes increasingly more important.

The author, during his course of instruction at the United States Naval Postgraduate School, was made aware of these facts and also of the lack of adequate data on local or point film heat transfer coefficients as applied to two-phase flow of steam and water.

Investigations of the subject of film heat transfer coefficients as applied to two-phase flow of steam-water were carried out at the United States Naval Postgraduate School in 1953 by Fisher and King (6) and in 1954 by Davis and Duacsek (4). These two investigations produced pertinent results and, with the encouragement of Professor E. E. Drucker, the author attempted to amplify and extend the range of data previously obtained in order to correlate optimum coefficients with quality of steam.

The experimental work of this thesis was carried out at the United States Naval Postgraduate School from March to May of 1955.

The writer wishes to express his appreciation of the encouragement, active assistance and cooperation of Professor E. E. Drucker in conducting the experimental work and in the preparation of this paper, and to Chief N. V. DuCette, USN; Chief F. H. Mietchen, USN; and J. H. Cox, EN-2 for their invaluable aid in the construction of the experimental set-up.

TABLE OF CONTENTS

LIST OF ILLUSTRATIONS

TABLE OF SYMBOLS AND ABBREVIATIONS

(Listed in the order of their use in the text)

A	- Area of heat transfer surface, square feet
C_p	- Specific heat at constant pressure, BTU/(lb)(deg F)
d	- Diameter, feet
G	- Mass velocity, lb/(hr)(sq ft of cross section)
h	- Coefficient of heat transfer between fluid and surface, BTU/(hr)(sq ft)(deg F)
k	- Coefficient of thermal conductivity, BTU/(hr)(ft)(deg F)
KW	- Kilowatt
L	- Heated length of tube, feet
μ	- Fluid Viscosity, lb/(hr)(ft)
p	- Total pressure, psia
π	- 3.1416
ρ	- Fluid density, lb/cu ft
q	- Rate of heat transfer, BTU/hr
r	- Radius, feet
t	- Temperature, degrees Fahrenheit
Δt	- Temperature difference, (deg F)
v	- Fluid velocity, ft/hr
Nu $\frac{hd}{k}$	- Nusselt Group, dimensionless
Pr $\frac{\mu c_p}{k}$	- Prandtl Group, dimensionless
Re $\frac{\rho v d}{\mu}$	- Reynolds Group, dimensionless

SUBSCRIPTS: d depth i inner o outer s surface

CHAPTER I

INTRODUCTION

Two-phase flow occurs in a wide range of applications in both forced and natural circulation flow. Despite this wide application, the basic heat-transfer and fluid-flow phenomena are not yet fully understood due to the large number of variables which must be incorporated in the anlysis.

In view of the increasing need for higher thermal efficiencies, it is felt that some re-evaluation of equipment design as it relates to heat transfer may bring about needed improvement. Accordingly, the need for investigation of the associated heat transfer variables is self-evident.

The objective of this thesis was to investigate the variation of the film heat transfer coefficient as associated with moisture content, mass velocity and heat flux. Primarily, the investigation was concerned with the variation of the film coefficient over a full range of moisture content; the other aforementioned variables being constant. Data was obtained for steam with a quality of 13% to 80% at a mass rate of 1270 pounds per hour and constant heat input. Time and equipment limitations were limiting factors in preventing further investigation at the lower and higher moisture content ranges. Time was also a limiting factor in the investigation of the coefficient variation with the mass velocity and heat flux variables. However, some data were obtained which may show a trend as related to these last two variables.

A secondary objective of this investigation was the visual observation of the two-phase flow phenomena and their correlation with associated heat transfer. These observations were carried out in conjunction with the primary objective.

The results of this investigation are presented as a function of moisture content of the two-phase mixture. Heat transfer coefficients relating to forced convection are usually correlated with respect to the local Nusselt, Reynolds and Prandtl numbers. Such correlation, however, is invariably for single phase flow and the lack of data on values of density, viscosity and thermal conductivity of wet steam precluded the computation of the above dimensionless quantities.

CHAPTER II

TWO-PHASE FLOW

Heat transfer in two-phase flow is important not only in vaporization phenomena inside tubes, but in many other applications as well.

While many data have been collected and published for vaporization inside tubes, very little seems to have been published with respect to the gas-liquid systems. From the work of McAdams and others (8), (9), who report local overall coefficients of heat transfer on vaporization inside tubes, it became apparent that in the preheating section, that is, before the actual boiling section, the film coefficient was usually higher than would follow from the Nusselt relation for single-phase flow in forced convection through a tube:

$$Nu = 0.024 Re^{0.8} Pr^{0.4}$$

This was presumed to be due to the fact that some vapor bubbles were formed in the preheating section and acted to decrease the thickness of the film and disrupt it locally and, upon moving into the stream, they condensed, thereby giving up their heat of condensation; the overall effect being to increase the film coefficient of heat transfer. From this it seems that a sound analysis of boiling inside tubes involves consideration of the variation in local conditions along the tube, including the relative fractions of vapor and liquid.

Several studies have now been reported for two-phase mixtures

of gas and liquid. Yoder and Dodge (12) measured film coefficients of Freon-12 boiling in a vertical tube and found a very significant change in the film coefficient with change in vapor content. Verschoor and Stemerding (11), in an air-water investigation, reported that the film coefficient reached a maximum at the change over from slug to annular flow. Bergelin (2) reported that if a gas and a liquid pass simultaneously in upward flow through a vertical tube then, as the gas'liquid ratio is increased, the following three flow pictures can be distinguished.

(a) bubble flow, when gas bubbles pass individually through the tube.

(b) slug flow, when alternatively slugs of gas and liquid pass through the tube, and

(c) annular flow, characterized by the fact that the liquid flows in an annulus along the tube wall, while the gas passes at a much higher velocity through the center of the tube.

Fisher and King (6) and others (4), (5) in reporting on high quality steam mixtures showed a marked increase in the local heat transfer coefficient as the moisture content increased.

CHAPTER III

EQUIPMENT

The experimental layout presented in the schematic diagram of Figure 1 was designed after taking into account the equipment limitations of past investigations, (4) and (6). A portion of the system may be seen in the photograph of Figure 2.

The equipment consisted essentially of a flow system in which steam was taken from a main steam line, through a throttling valve, a Centrifix Separator, passed vertically downward through 15 feet of $1\frac{1}{2}$ inch piping and then vertically upward to the test section. Saturated water was spray injected into the steam on the lower end of the upward flow section which continued as $1\frac{1}{2}$ inch pipe for eight feet. The purpose of this loop was to give the steam-water mixture a long straight path, prior to entering the test section, in order to achieve equilibrium.

After passing upward through the eight feet of $1\frac{1}{2}$ inch pipe the flow channel was reduced to one-half inch diameter and the flow continued upward through three feet of flexible metal hose, a sight section to observe flow characteristics, another three feet of flexible metal hose and into the test section. The purpose of this flexible metal hose was to allow for expansion of the piping and dampen vibrations in order to prevent the test section from warping and the glass tube in the sight section from breaking.

At the exit of the test section the flow was reversed and passed vertically downward to a Centrifix Type RA separator. From the

steam outlet of the Centrifix the separated steam was passed through a needle control valve to a Fisher and Porter Co. Series 50 Flowrator meter, through 16 feet of 3/4 inch pipe, 35 feet of 1½ inch pipe, 7 feet of 2 inch pipe and into a condenser operated at atmospheric pressure. This piping, which had been previously installed (4), facilitated the attainment of mass rates up to 1500 pounds per hour through the test section. To prevent "spill-over" of water into the dry steam line of the separator, a small level tank and gage glass were installed in parallel with the separator, proper flow rate being maintained by a needle valve on the drain line. Immediately following this needle valve was mounted a Fisher and Porter Series 700 Flowrator Meter. After passing through the meter the water was discharged directly into the condenser.

The moisture content of the steam was determined by an Ellison U-Path Steam Calorimeter immediately following the separator. It was found that the moisture content of the steam reached a maximum of 1% at an inlet moisture content of 86%. Below an inlet moisture content of 70% the moisture content of the exit steam from the separator was found to be negligible.

Steam supply came from a Babcock and Wilcox FM Boiler at an operating pressure of 200 psig.

Water supply was effected by means of the condensate which was fed into a supply tank by means of the condenser condensate pump. This supply tank was mounted on an elevated platform in order to allow for future installation of a Duolite deionizer which was not available for this investigation.

Water was injected by means of a centrifugal vane type pump which took suction from the supply tank and discharged through a needle valve, two Schutte and Koerting Company jet type steam heaters mounted in series and thence injected into the mixing line by means of spray nozzle. The jet type heaters were capable of heating up to 1100 pounds of water per hour to saturation temperature, steam supply to the heaters being insufficient for higher rates.

The test section, details of which are shown in Figure 3, was manufactured from a tellurium-copper bar (99.35% Cu, 0.65% Te). This particular composition was selected because of its improved machining properties over that of pure copper.

Details of the thermocouple installations are shown in Figure 4. The thermocouples installed in the walls of the test section were constructed of No. 30 gauge copper-constantin wire. The thermocouple junctions were welded in a mercury arc and hand finished with fine emery to give a dimension of 0.015 inch between the tip of the bead and the junction. A bare thermocouple junction was projected into the center of the stream at either end of the test section. Pressure taps were made as shown in Figure 3.

Heat supply to the test section consisted of four independent heating elements containing 68 feet of No. 17 Nichrome V wire wound around each of the four sections in a single layer. Each heating element completely covered each 6 inch section. A thin layer of mica was wrapped around the bare metal cylinder prior to winding the wire. Power supply to the heaters was controlled by four two-gang Variac assemblies and measured by portable wattmeters.

Heat insulation of the test section consisted of wrapping several layers of glass tape on the outside of the heating coils, covering this with three layers of asbestos tape, fitting a $2\frac{1}{2}$ inch layer of magnesia brick over this and covering the whole with one-half inch of wet magnesia mix. The Centrifix and all piping up to the steam flow meter were similarly insulated. Heat loss was considered negligible through this insulation.

All components of the system were constructed, as far as was practicable, of non-ferrous materials.

CHAPTER IV

OPERATING PROCEDURE

At the end of each operating period, the system was drained and left in a dry condition in order to prevent accumulation of impurities. Prior to each day's operations, the system was flushed with dry steam and allowed to drain, after which, the supply tank was filled with clean water from accumulated condensate. This complete operation required approximately two hours.

The operating variables associated with the apparatus were pressure, heat flux, flow rate and moisture content. The inlet pressure to the test section was maintained constant at 150 psig for all runs. The heat input per heater was limited to 1.5 KW for all but two runs.

A total of 27 runs were made, of which, 25 were considered to be satisfactory. All but four runs were made at a constant mass rate of 1270 pounds per hour with an average variation of 5 pounds per hour.

The establishment of proper flow rates and moisture content was facilitated by the use of the flowrator meters. However, very delicate adjustment of the control valves was required in order to maintain the proper flow rates of both the steam and the water.

The approach to steady-state conditions was determined by noting the change in the millivolt readings of the thermocouples in the wall of the test section. It normally took 30 minutes for these readings to become steady after the system had been warmed up and the proper flow rate and moisture content had become steady. The

overall time required to make one run was approximately 2 hours.

It was found, due to insufficient steam supply to the jet heaters, that moisture content variations greater than 86% were impracticable at the high flow rate selected for the majority of the runs. It was also found that moisture content variations below 20% were impracticable due to the inherent flow capacity of the water flowrator meter. A change was later made in the drain line of the Centrifix separator to allow the use of a weigh tank in determining these lower moisture contents. However, the limited time available precluded the procurement of this data.

For each run, thermocouple millivolt readings, pressure and pressure drops, flow rates and heat input were recorded.

Standard Fisher and Porter Company correction curves were used in determining the true flow rates through the flowrator meters.

CHAPTER V

METHOD OF CALCULATION

The film heat transfer coefficient, h, is defined as the proportionality factor in Newton's law of cooling:

$$dq = hdA(t_s - t)$$

where dq is the local rate of heat transfer through a surface element dA.

In this investigation, heat loss was assumed to be negligible and the total heat generated in the heating coils was assumed to be transferred to the fluid. The rate of energy supplied was measured by means of wattmeters.

The thermocouples in the test section wall were calibrated by passing dry steam through the test section with no heat input and allowing the system to reach equilibrium. This calibration is shown in Figure 5.

Knowing the temperature at the thermocouple junctions, the surface temperature, t_s, was determined by using the Fourier conduction equation for steady state heat transmission:

$$q = - kA \frac{dt}{dx}$$

For the circular tube, this equation was integrated to give:

$$q = \frac{2\pi k L \Delta T}{\ln r_o/r_i}$$

The coefficient of thermal conductivity, k, was extrapolated from the temperature variation curve for pure copper knowing the value of k for the test section at a specified temperature. For a power input of 1.5 KW per coil, the temperature drop due to conduction was calculated to be 2.60°F.

Owing to the construction of the test section and to allow for end effects, it was decided to average the thermocouple values of the two center sections of the test section in obtaining one average surface temperature, t_s, for these two sections. The maximum variation in these thermocouple readings seldom exceeded 0.5°F.

It was determined, at the mass velocities and rate of heat transfer used, that any change in moisture content of the mixture while passing through the test section was negligible. It was also determined that any change in moisture content between the test section and flow meters was negligible. Accordingly, the moisture content and mass rate of flow were determined from the corrected readings of the flow meters.

The temperature and pressure of the stream were measured at the entrance to the test section, with pressure drops along the test section being measured through means of a differential manometer. The temperature of the steam at the point for which the average surface temperature, t_s, was calculated was, therefore, determined by applying the pressure drop to the corresponding saturation pressure indicated by entrance temperature.

The temperature difference, as indicated in the convection equation, varied from 6.91°F to 17.32°F, depending on the moisture

content, mass velocity, and heat input.

CHAPTER VI

CONCLUSIONS AND RESULTS

Twenty-seven runs were made, of which 25 were considered to be successful. The remaining two runs were eliminated because the data indicated that steady state conditions had not yet been reached.

The data for the heat transfer coefficient as a function of the percent moisture are shown in Figure 6. This figure shows one continuous curve for a single mass velocity from 19.9% to 86.9% moisture. The Nusselt relation for single-phase flow was used in calculating the coefficient for saturated water at this mass velocity and shows very good agreement with extrapolation of the curve from 86.9% to 100% moisture. Data are also shown in the figure for variation of the heat transfer coefficient with mass velocity at 50% moisture, also shown in Figure 7, and with heat input at 38% moisture.

Data taken from the faired-in curves presented by Fisher and King (6) are also shown in Figure 6. Any correlation of the two sets of data would indicate an optimum heat transfer coefficient falling in the range of 7% to 20% moisture content, depending on the mass velocity, with heat input showing relatively small effect.

It could be presumed from the data as presented in Figure 6 that, for any given moisture content, the heat transfer coefficient would vary inversely with mass velocity at the lower moisture contents on one side of the optimum and directly with the mass velocity at the higher moisture contents on the other side of the optimum. This could possibly give rise to a family of peaks within a narrow range

of optimum heat transfer coefficients and moisture content. However, such a presumption, without further verification, could not be considered reliable.

Figure 8 shows the variation of the pressure drop component with percent moisture. Search of existing published data showed nothing in this range, making a comparison impossible. However, Dengler (5) shows a sharp rise in the pressure drop component in the range 3% to 10% vapor by weight.

Figure 9 shows the correlation of the type of flow visually observed as related to heat transfer coefficient versus percent moisture content. The types of flow observed in the range of 20% to 87% moisture were:

(a) well mixed fog flow with no annular flow observed up to approximately 30% moisture content.

(b) well mixed fog flow with a slight annular flow beginning at approximately 30% moisture content.

(c) well mixed fog flow and light annular flow with an occasional more dense fog mixture in the same pattern as solid slug flow beginning at approximately 40% moisture.

(d) medium annular flow with a decrease in the fog density in the center of the channel. Fog slugs absent beginning at approximately 55% moisture.

(e) heavy annular flow at 58% moisture. Fog flow not noticeable.

(f) very heavy turbulent annular flow beginning at

approximately 65% moisture. Breaking into occasional slug.

(g) slug flow beginning at 68% moisture.

It will be observed from Figure 9 that some correlation may be made between the slope of the curve and the change in the type of flow. It is felt that further investigation of this phenomena is warranted in regard to correlation of heat transfer coefficient, mass velocity and percent moisture.

The results of this investigation tend to show the existance of an optimum heat transfer coefficient in a narrow range of moisture content for a given mass velocity. Further investigation is certainly justified in regard to possible improved design of heat transfer equipment.

BIBLIOGRAPHY

1.	Baker, H.D.; Ryder, E.A.; and Baker, N.H.	TEMPERATURE MEASUREMENT IN ENGINEERING, John Wiley & Sons, Inc., New York, 1953
2.	Bergelin, O.P.	HEAT TRANSFER AND FLUID MECHANICS INSTITUTE, 1949 (University of Calif., Berkeley)
3.	Brown, G.	HEAT TRANSMISSION BY CONDENSATION OF STEAM ON A SPRAY OF WATER DROPS., I.M.E. and A.S.M.E. Proceedings, General Discussion on Heat Transfer, 49-52, September, 1951
4.	Davis, C.S., Jr., and Duacsek, A. W.	HEAT TRANSFER COEFFICIENTS OF STEAM WATER MIXTURES, 1954 (U. S. Naval Postgraduate School)
5.	Dengler, Carl E.	HEAT TRANSFER AND PRESSURE DROP FOR EVAPORATION OF WATER IN A VERTICAL TUBE, 1953., (Massachusetts Institute of Technology)
6.	Fisher, L. W., and King, J. M.	HEAT TRANSFER COEFFICIENTS OF STEAM WATER MIXTURES, 1953 (U. S. Naval Postgraduate School)
7.	Griffiths, Ezer.	METHODS OF MEASURING TEMPERATURE, J. B. Lippincott Co., Phil.
8.	McAdams, W. H.	HEAT TRANSMISSION, (2nd Edition), McGraw-Hill, New York, 1942
9.	McAdams, W. H.; Drexel, R. E., and Goldey, R.H.	TRANSACTIONS, A.S.M.E., 67:613, 1945
10.	McAdams, W. H.; Kennel, W. E., and Alloms, J. N.	HEAT TRANSFER TO SUPERHEATED STEAM AT HIGH PRESSURES., Transactions, A.S.M.E., 72:421-430, 1950
11.	Verschoor, H., and Stemerding, S.	HEAT TRANSFER IN TWO-PHASE FLOW. I.M.E. and A.S.M.E. Proceedings, General Discussion on Heat Transfer, 201-204, September 1951

12. Yoder, R. J., and Dodge, B.F. HEAT TRANSFER COEFFICIENTS OF BOILING Freon-12, I.M.E. and A.S.M.E. Proceedings, General Discussion on Heat Transfer, 15-19, September, 1951

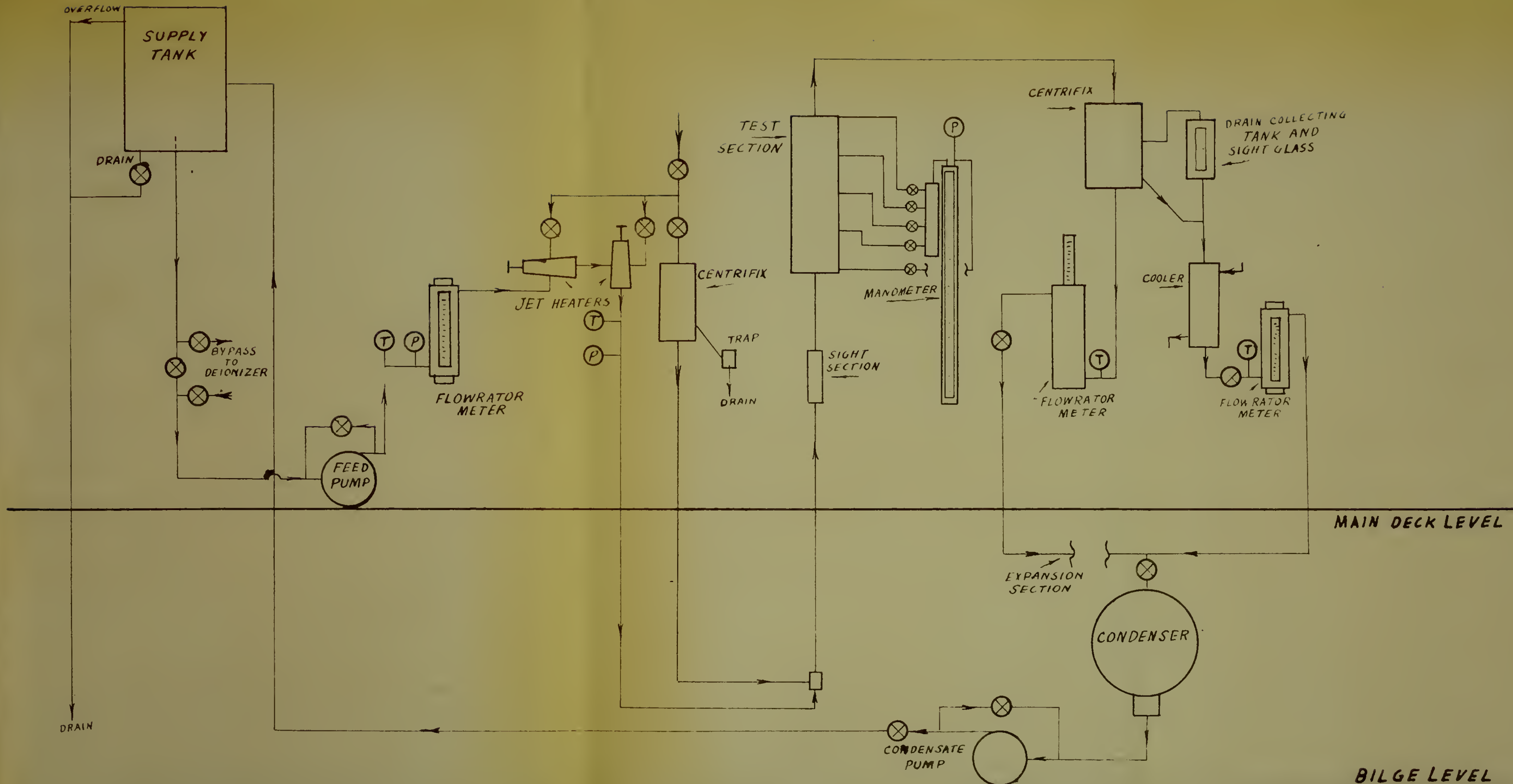


FIGURE 1 - SCHEMATIC FLOW DIAGRAM

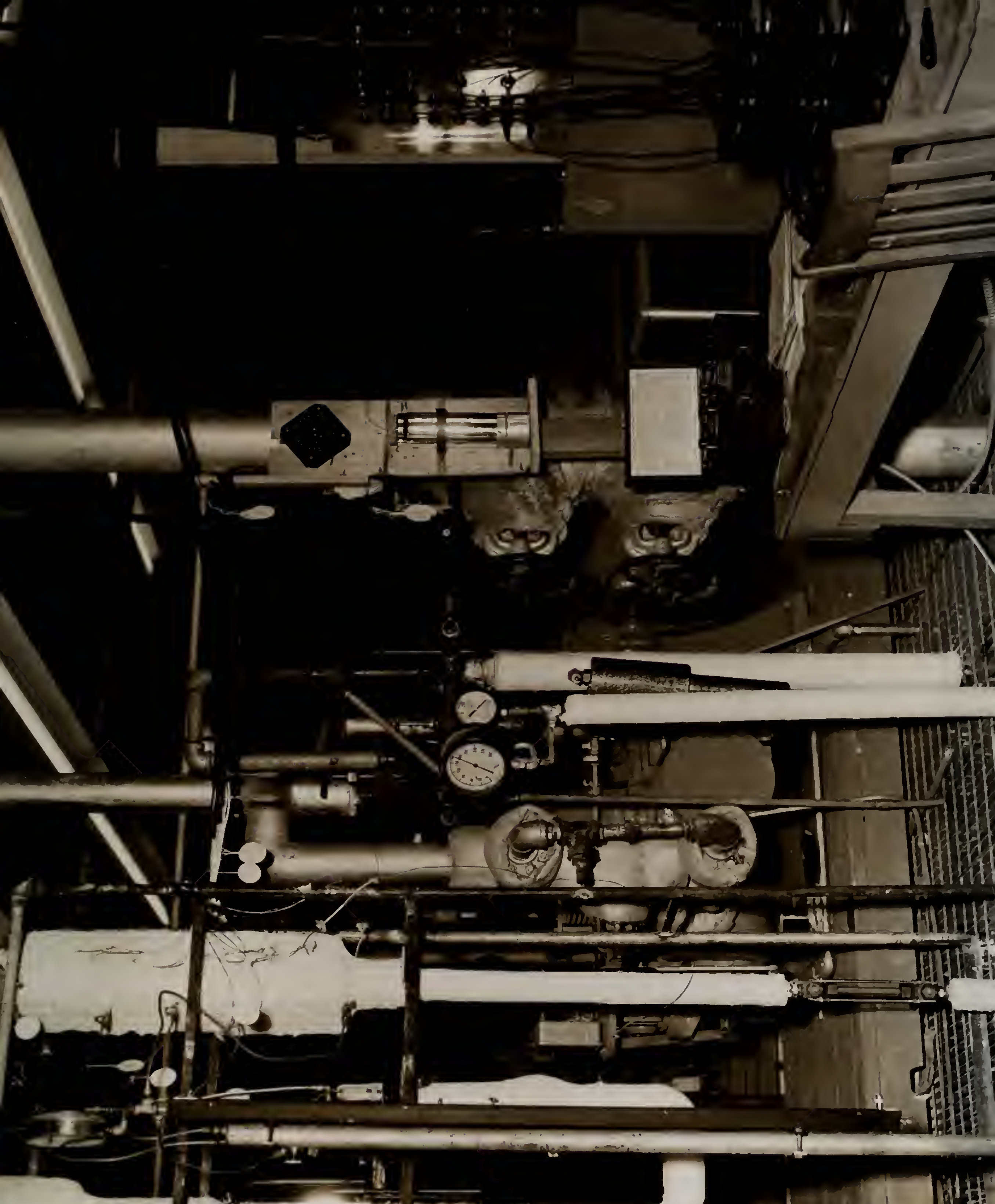
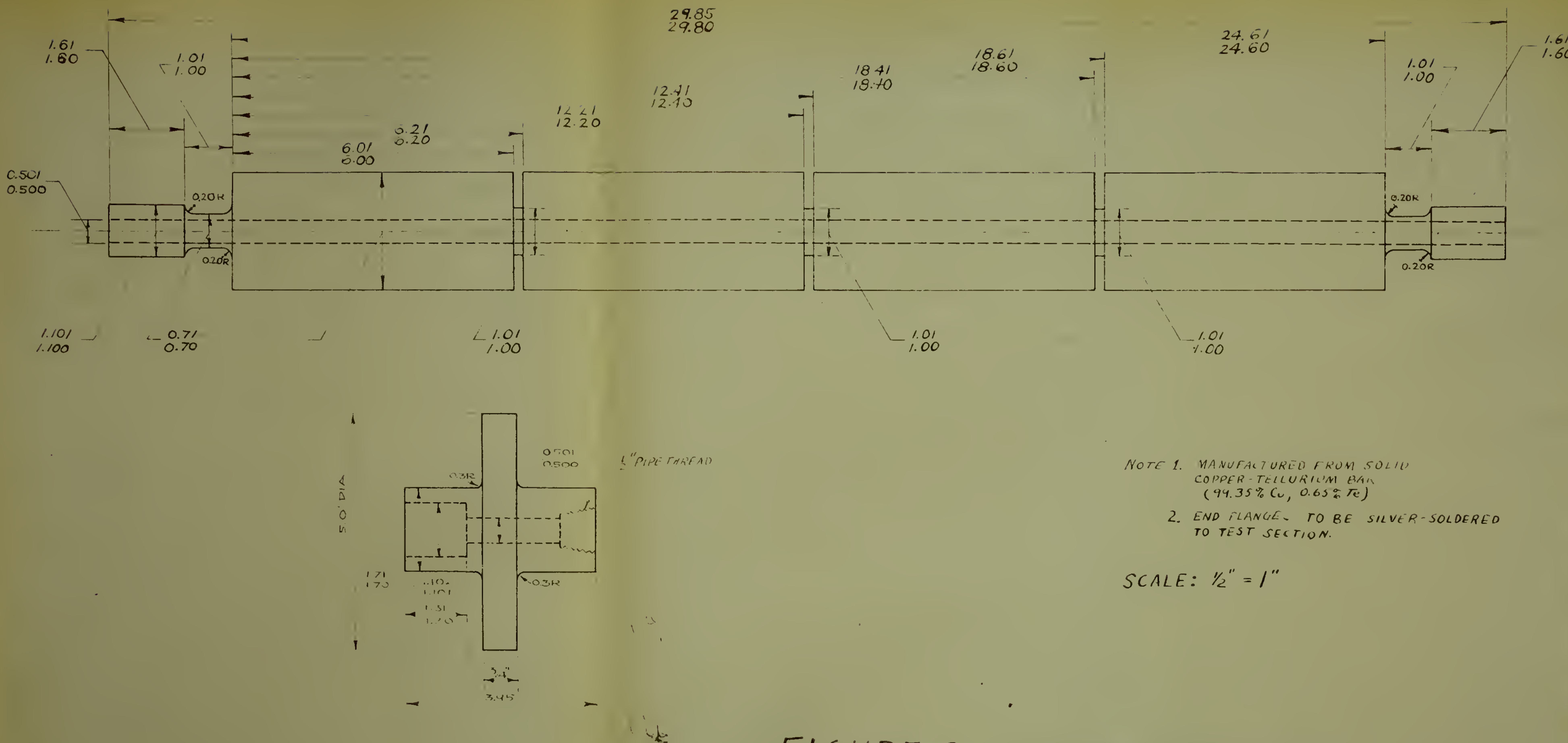


FIGURE 2



NOTE 1. MANUFACTURED FROM SOLID COPPER-TELLURIUM BAR (99.35% Cu, 0.65% Te)

2. END FLANGES TO BE SILVER-SOLDERED TO TEST SECTION.

SCALE: ½" = 1"

FIGURE 3 - TEST SECTION DETAILS

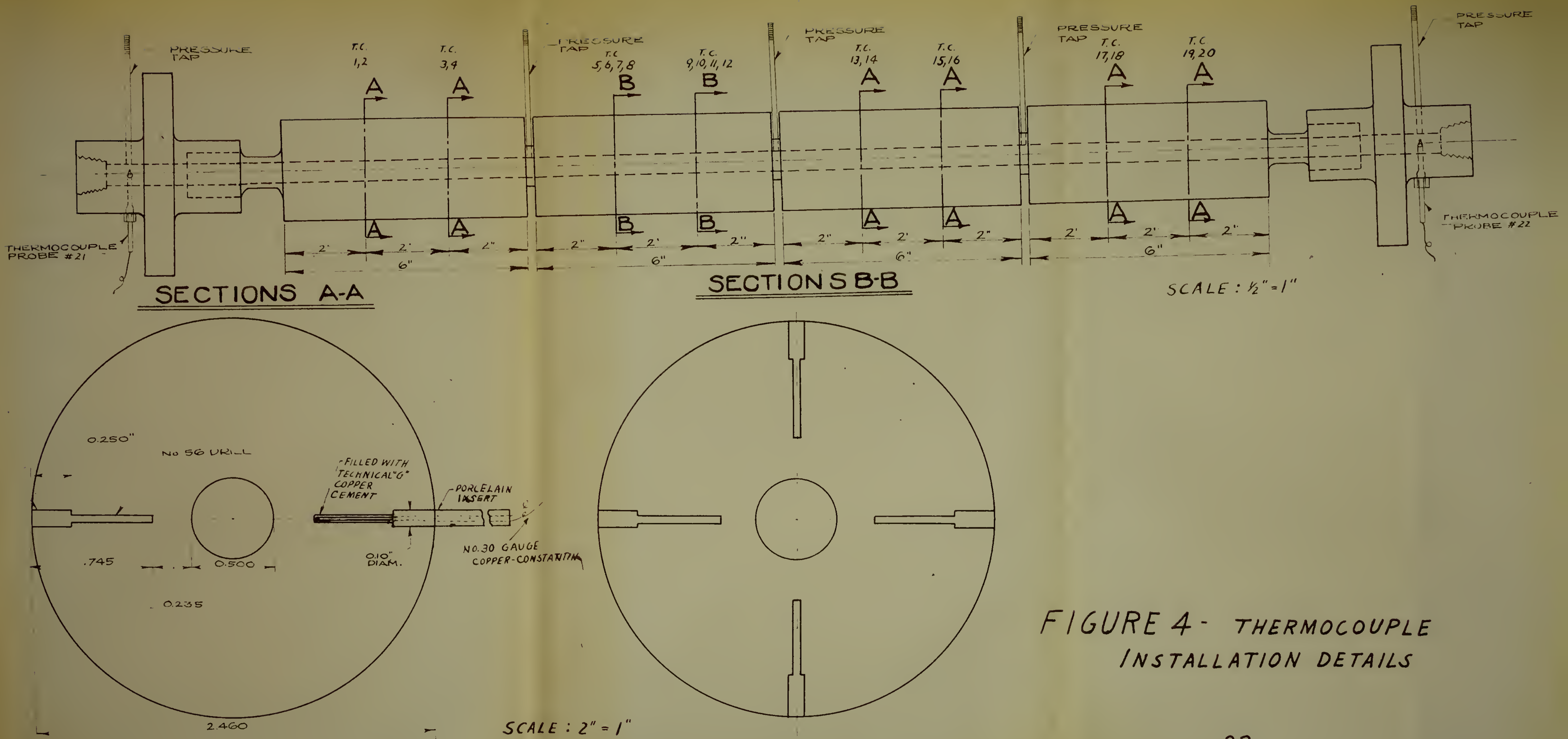


FIGURE 4 - THERMOCOUPLE INSTALLATION DETAILS

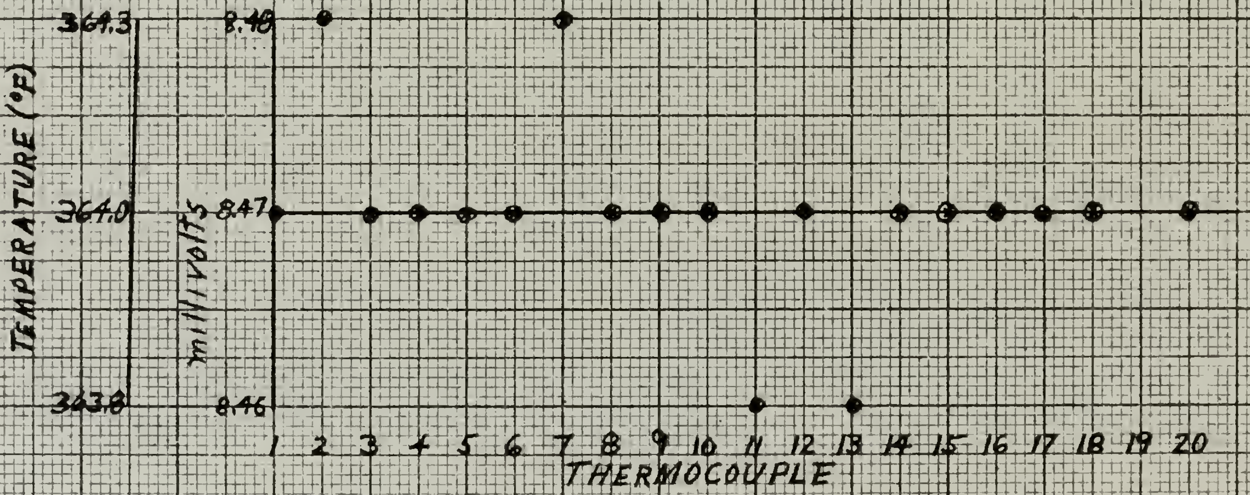
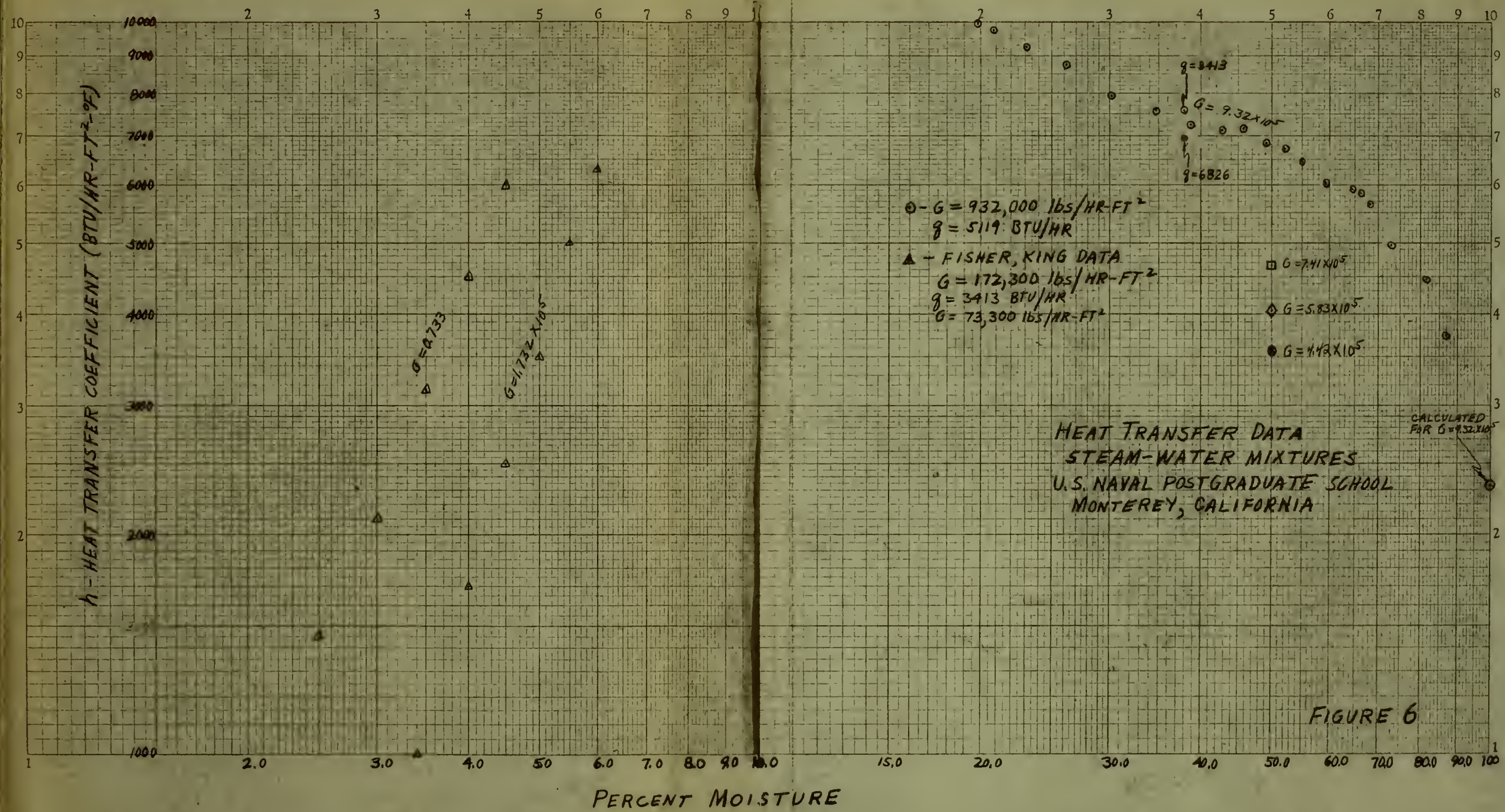
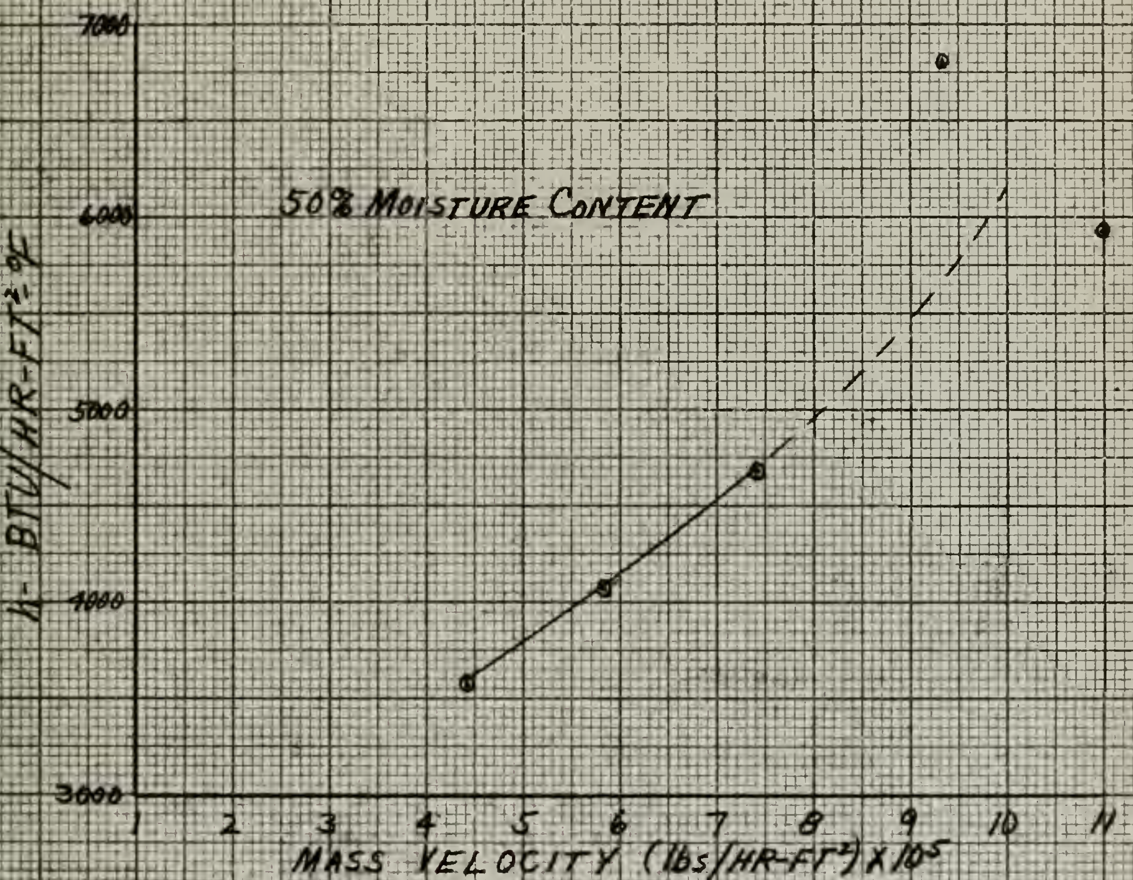


FIGURE 5 - THERMOCOUPLE CALIBRATION

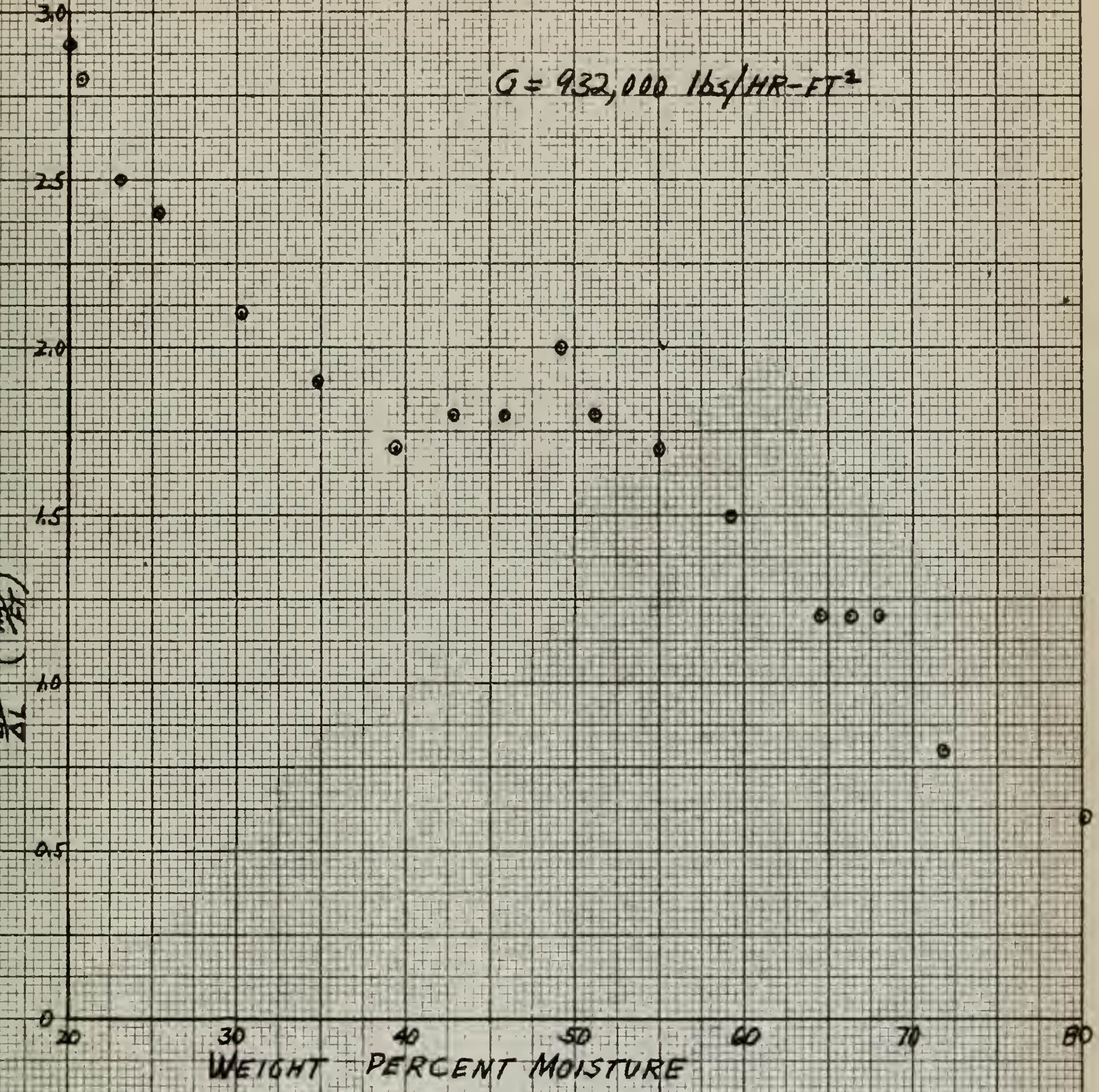
CONSTANT STEAM INPUT
NO HEAT INPUT
EQUILIBRIUM READINGS SHOWN
COPPER - CONSTANTIN
38 CALIBRATION, 32°F REFERENCE


h - HEAT TRANSFER COEFFICIENT (BTU/HR-FT²-°F)
PERCENT MOISTURE
⊙ - G = 932,000 lbs/HR-FT²
q = 5119 BTU/HR
▲ - FISHER, KING DATA
G = 172,300 lbs/HR-FT²
q = 3413 BTU/HR
G = 73,300 lbs/HR-FT²
q = 3413
G = 9.32×10⁵
q = 6826
G = 7.41×10⁵
G = 5.83×10⁵
G = 4.42×10⁵
q = 4733
G = 1.732×10⁵
CALCULATED FOR G = 9.32×10⁵
HEAT TRANSFER DATA
STEAM-WATER MIXTURES
U.S. NAVAL POSTGRADUATE SCHOOL
MONTEREY, CALIFORNIA
FIGURE 6



HEAT TRANSFER DATA
STEAM-WATER MIXTURES
U.S. NAVAL POSTGRADUATE SCHOOL
MONTEREY, CALIFORNIA

FIGURE 7



HEAT TRANSFER DATA
PRESSURE DROP COMPONENT
VS.
WT. PER CENT MOISTURE

U.S. NAVAL POST GRADUATE SCHOOL
MONTEREY, CALIFORNIA

FIGURE 8

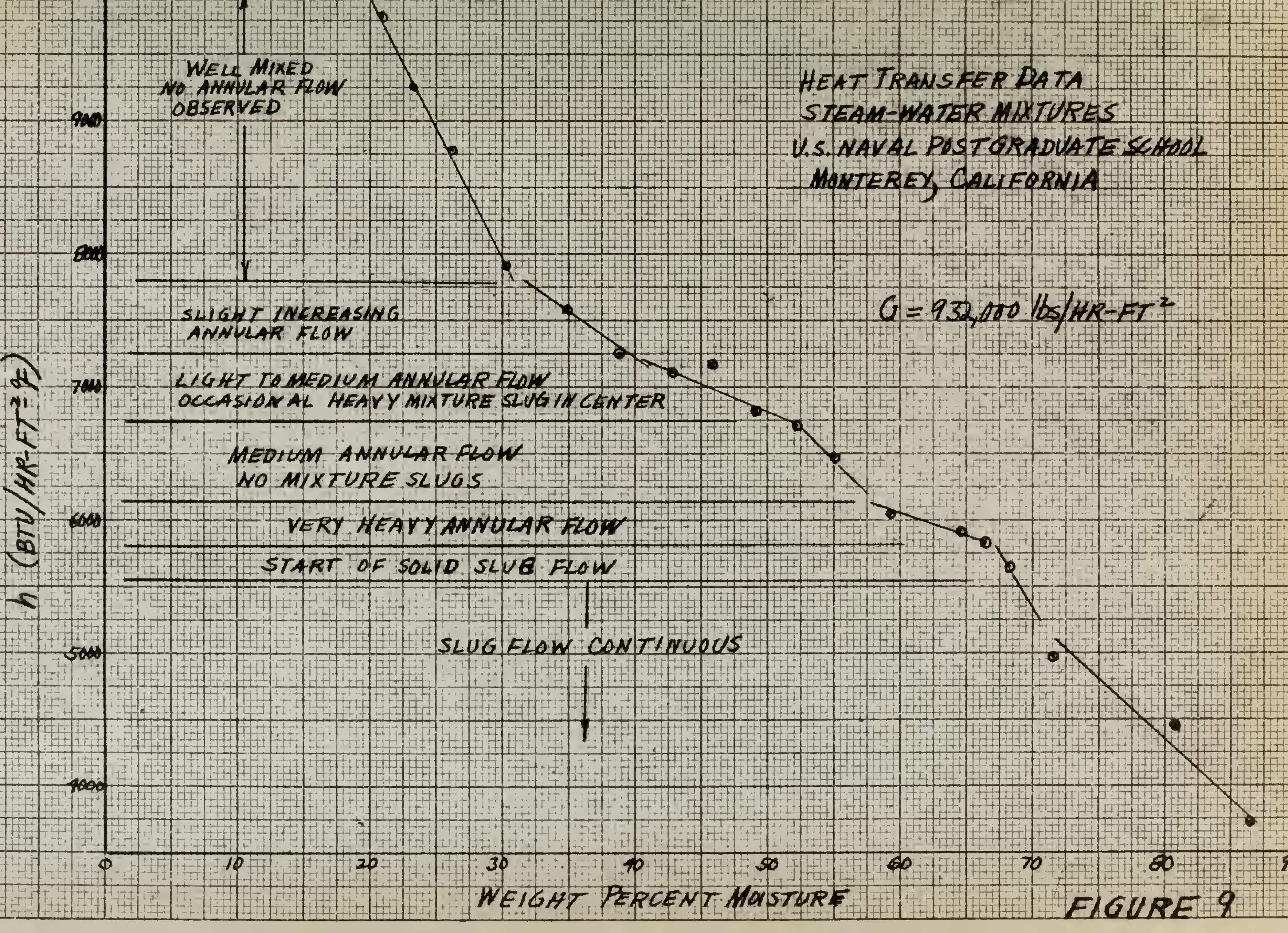

HEAT TRANSFER DATA
STEAM-WATER MIXTURES
U.S. NAVAL POSTGRADUATE SCHOOL
MONTEREY, CALIFORNIA
G = 932,000 lbs/HR-FT²
WELL MIXED
NO ANNULAR FLOW
OBSERVED
SLIGHT INCREASING
ANNULAR FLOW
LIGHT TO MEDIUM ANNULAR FLOW
OCCASIONAL HEAVY MIXTURE SLUG IN CENTER
MEDIUM ANNULAR FLOW
NO MIXTURE SLUGS
VERY HEAVY ANNULAR FLOW
START OF SOLID SLUB FLOW
SLUG FLOW CONTINUOUS
h (BTU/HR-FT²-°F)
9000
8000
7000
6000
5000
4000
0
10
20
30
40
50
60
70
80
WEIGHT PERCENT MOISTURE

FIGURE 9

Thesis

28467

N364

Nelson
Heat transfer coefficients
of steam water mixtures.